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A Simplified Model for Assessing Improvement Potential of Air-To-Air Conditioners and Heat Pumps

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ABSTRACT

Many countries have adopted energy efficiency regulations for mini-split type air conditioners in the world. For these products, energy efficiency metrics is moving to seasonal performance evaluation. Many countries have already changed their metrics and many others are planning to customize the ISO 16358:2013 CSPF (Cooling Seasonal Performance Factor) and HSPF (Heating Seasonal Performance Factor) indicators to their own climate and use of air conditioners.

To build policies to justify the need for implementing MEPS (Minimum Energy Performance Standard) and energy label programs, technico-economic model is required. A standard product is to be modeled as well as potential improvement options. The complexity of the task increases significantly when shifting from one or two test point (rated EER or COP) to several test points to compute SEER and SCOP.

In this context, we propose a simple model to evaluate the various EER and COP values for different part load conditions in such a regulatory context. The model is first described; governing equations by component are given, as well as technical parameters required as input. The model is validated using several different units for which EER and COP at part load conditions according EU seasonal performance standard are known. Potential of the model to simulate improvement options in view of regulatory development is discussed in conclusion, as well as potential improvements.

Keywords:

Mini-Split; Energy Efficiency Regulation; Seasonal Performances; Modelling; Thermodynamic; Air Conditioning; Compressor; Heat Exchangers

1. INTRODUCTION

Air conditioners represent a major energy end use in many countries, and contribute significantly in the total electricity consumed in buildings.
Many seasonal performance metrics have been developed recently to describe the performance of cooling and heating of air conditioners and heat pumps and they provide a better basis to compare energy efficiency of products than nominal standard rating conditions alone. But these metrics are costly in terms of the number of points at which the product must be evaluated.
In many cases these different metrics cannot be compared directly with each other. In this case, energy performance modeling is needed to provide an estimate of energy efficiency indicator of the same products for different metrics. Also, the impact of improvement options depends on the imposed evaluation metrics. Those two exercises, benchmarking of different regional policies and study of improvement potential are required for policy design. To do so, it is necessary to have a simple model, easy to manipulate, which allows to simulate quickly several points and to assess the efficiency gain of each improvement option.
The simple model we present here is based on thermodynamics and thermal transfer laws, using however, simplifying assumptions, some of which have already been verified in the literature, while others are presented and discussed here.

2. MODELING ENERGY PERFORMANCE

2.1 Calculation methods of SEER and SCOP in the European Regulation

Regulation (EU) N° 206/2012 [1] provides part load conditions and calculation methods for calculating the Seasonal Energy Efficiency Ratio (SEER) and Seasonal Coefficient of Performance (SCOP) of air conditioners and heat pump units when they are used to fulfill building cooling and heating demands. The part load conditions for air-to-air units in cooling/heating mode for determining the SEER/SCOP are given in the following tables:

Table 1: Part load conditions for reference SEER and reference SEERon calculation of air-to-air units

	Part load ratio	Part load ratio %	Outdoor air dry bulb temperature °C	Indoor air dry bulb (wet bulb) temperatures °C
A	(35-16)/($T_{designC}$ -16)	100	35	27(19)
B	(30-16)/($T_{designC}$ -16)	74	30	27(19)
C	(25-16)/($T_{designC}$ -16)	47	25	27(19)
D	(20-16)/($T_{designC}$ -16)	21	20	27(19)

Table 2: Part load conditions for reference SCOP calculation of air-to-air units for average heating season

	Average heating season		Outdoor air dry bulb (wet bulb) temperatures °C	Indoor air dry bulb temperature °C
	Part load ratio	Part load ratio %		
A	(-7-16)/($T_{designH}$ -16)	88	-7(-8)	20
B	(+2-16)/($T_{designH}$ -16)	54	2(1)	20
C	(+7-16)/($T_{designH}$ -16)	35	7(6)	20
D	(+12-16)/($T_{designH}$ -16)	15	12(11)	20
E	(T_{OL}-16)/($T_{designH}$ -16)		T_{OL}	20
F	($T_{bivalent}$-16)/($T_{designH}$ -16)		$T_{bivalent}$	20

Operation limit temperature (T_{OL}) is the lowest outdoor temperature at which the unit can still deliver heating capacity, as declared by the manufacturer. Below this temperature, the heat pump will not be able to deliver any heating capacity.

Bivalent temperature ($T_{bivalent}$) is defined as the lowest outdoor temperature point at which the unit is declared to have a capacity able to meet 100 % of the heating load. Below this point, the unit may still deliver capacity, but additional back up heating is necessary to fulfill the full heating load.

To compute SEERon and SEER, the formulas to be applied are:

$$SEER_{on} = \frac{\sum_{j=1}^{n} h_j \times P_c(T_j)}{\sum_{j=1}^{n} h_j \times \frac{P_c(T_j)}{EER(T_j)}}$$

$$SEER = \frac{P_{designC} \times H_{CE}}{\frac{Q_C}{SEER_{on}} + P_{TO} \times H_{TO} + P_{SB} \times H_{SB} + P_{CK} \times H_{CK} + P_{OFF} \times H_{OFF}}$$

The reference SCOP is defined as reference annual heating demand divided by the annual electricity consumption (the annual electricity consumption includes the power consumption during active mode, thermostat-off mode, standby mode, off mode and that of the crankcase heater), to compute SCOPon and SCOP, the formulas to be applied are:

$$SCOP_{on} = \frac{\sum_{j=1}^{n} h_j \times P_H(T_j)}{\sum_{j=1}^{n} h_j \times \left(\frac{P_H(T_j) - elbu(T_j)}{COP_{bin}(T_j)} + elbu(T_j) \right)}$$

$$SCOP = \frac{P_{designH} \times H_{HE}}{\dfrac{Q_H}{SCOP_{on}} + P_{TO} \times H_{TO} + P_{SB} \times H_{SB} + P_{CK} \times H_{CK} + P_{OFF} \times H_{OFF}}$$

where:

- j : the bin number / n : the amount of bins
- T_j : the bin temperature
- h_j : the number of bin hours occurring at the corresponding temperature T_j
- Q_C (Q_H) : The reference annual cooling (heating) demand, $Q_C = P_{designC} \times H_{ce}$ and $Q_H = P_{designH} \times H_{he}$, expressed in kW
- P_{TO}, P_{SB}, P_{CK}, P_{OFF} : the electricity consumption during respectively thermostat off mode, standby mode, crankcase heater mode and off mode, [kW]
- H_{TO}, H_{SB}, H_{CK}, $H_{OFF,}$ H_{CE} : the number of hours the unit is considered to work in respectively thermostat off mode, standby mode, crankcase heater mode, off mode and equivalent mode hours for cooling
- $P_C(T_j)$ ($P_H(T_j)$): is the heating (heating) demand of the building for the corresponding temperature T_j, [kW]
- $EER(T_j)$ ($COP_{bin}(T_j)$) : the EER (COP) values of the unit for the corresponding temperature Tj in cooling (heating) mode.
- $T_{designC}/T_{designH}$:Reference design temperature conditions for cooling/heating (TdesignH depend of climate conditions)
- H_{ce} is the numbers of equivalent heating hours (for the average, warmer and colder reference heating seasons types of units are based on occupancy scenarios for certain types of buildings and a climate bin method)
- $P_{designC}$ / $P_{designH}$: full load cooling/ heating (climate dependent for heating)
- $Elbu(T_j)$ is the required capacity of an electric backup heater for the corresponding temperature T_j, [kW]

The $COP_{bin}(T_j)$ ($EER_{bin}(T_j)$) values and heating (cooling) capacity values at each bin are determined via interpolation of the $COP_{bin}(T_j)$ ($EER_{bin}(T_j)$) and capacity values at part load conditions A, B, C, D, E and F (A, B, C and D for cooling) where applicable. Interpolation is done between the $COP_{bin}(T_j)$s ($EER_{bin}(T_j)$s) and capacities of the 2 closest part load conditions. In cooling mode, for capacities required at ambient larger than 35 °C (above A), it supposed the capacity required can be reached and EER_A value is used for these points. For capacities at bin corresponding to temperatures below 20 °C (D point), capacity and EER of D point are used and a cycling coefficient is applied.

In heating mode, the COPbin(Tj)s values and capacity values for part load conditions above D are extrapolated from the $COP_{bin}(T_j)$s values and capacity values at part load conditions C and D. If the capacity of the heat pump is lower than the value of P_H(Tj) (at low ambient), correction needs to be made for the missing capacity with an electric back up heater with a COP of 1.

At low loads in both cooling and heating mode, units may have difficulties to reach the low capacities required (point D in both modes, sometimes also point C in cooling mode); in that case, the capacity declared supposedly cannot reach the required capacity and a Cd coefficient of 0.25 (linear EER/COP degradation factor with load ratio) is used to correct the EER or COP of the unit according to standard EN14825:2016 [2].

In conclusion, EER and COP depend both on the cooling/heating capacities of the unit and on the compressor electricity consumption. Capacities are imposed by the testing points in the metric (declared capacities) and the choice of design parameters: $P_{designC}$ (corresponding to $T_{designC}$) in cooling mode, and respectively in heating mode $P_{designH}$, and P_{biv} (capacity corresponding to T_{biv}). These parameters with other general design parameters (as the air flow rate, compressor efficiency…) are known for most products. The present simplified model, with these main characteristic parameters, allows having an idea about the other unknown design parameters, which could be useful to reassess energy performance under other conditions for all imposed points by the performance evaluation metric.

2.2 General outline of the simplified model

The model is designed and verified, based on the performance evaluation conditions imposed by the European regulations (European Standard EN 14825:2016, EU harmonized standard for Regulation n°206/2012). The initial purpose was to develop a simple thermodynamic based evaluation tool to compute the impact of improvement options on the energy efficiency indicators of products, in particular, for the options that regard compressor performance and heat exchanger effectiveness.

For the European regulation, SEER calculation requires to model the performance of the EER values (for reduced outdoor temperature and capacity ratios) at the following test points: A (100%/35 °C), B (74%/30 °C), C (47 %/25 °C) and D (21 %/20 °C). In the same manner, SCOP calculation requires to compute at least 5 performance points for varied outdoor temperature and part load conditions: F (-10 °C/max declared capacity), A (88%/-7 °C), B (2 °C/54 %), C (7 °C/35 %) and D (12 °C/15%).

2.2.1 Evaporating and condensing temperature estimates

Evaporation temperature (T_{Evap}) in cooling mode:

For the evaporator in cooling mode, the cooling capacity to be exchanged is known. There are two distinct situations to compute the evaporating temperature:
Case 1: for EER_A (100% part load/ 35 °C outdoor temperature) and in most cases for EER_B (74% part load/ 30 °C outdoor temperature) test conditions, there is dehumidification. In that case, the heat exchanger capacity is computed from an assumed heat exchanger effectiveness value (also called bypass factor for a coil with dehumidification) and a given air flow rate. Cooling capacity is decreased by the fan motor power (supposing that all motor losses convert to heat in the air stream and that useful fan energy converts to pressure losses and then to heat in the air stream ultimately). Refrigerant fluid evaporating temperature (T_{Evap}) is identified by iteration so that the sum of the sensible and latent capacities reaches the cooling output of the simulated point.
Case 2: for EER_c (47% part load/ 25 °C outdoor temperature) and EER_D (21% part load/ 20 °C outdoor temperature), there is no dehumidification. In that case, (T_{Evap}) is identified by iteratively equalizing two DTLM (the logarithmic mean temperature difference between air and refrigerant) values computed with the help of the equations below:

$$DTLM_1 = Q_{cooling} / UA_{ev}$$

$$DTLM_2 = \frac{(T_{a,i} - T_{ev}) - (T_{a,o} - T_{ev})}{Ln\left\{ (T_{a,i} - T_{ev}) / (T_{a,o} - T_{ev}) \right\}}$$

with : $UA_{ev} = NUT_{ev} \times mCp_{ev}$ and $NUT_{ev} = ln\left(1 / 1 - \varepsilon_{ev} \right)$

where:

- $Q_{cooling}$: cooling capacity to be exchanged at evaporator, [kW]
- UA: global heat exchange coefficient of the heat exchanger, [W/K]
- m: air flow rate, [kg/s]
- C_p: air specific heat at constant pressure, [J/kg/K]
- $T_{a,i}$ ($T_{a,o}$) : evaporator inlet (outlet) air temperature
- T_{Evap}: Refrigerant fluid evaporating temperature
- NUT: number of unit transfer (ratio of UA to mCp). NUT_A refers to the reference point used to fix UA, in that case point A (100% part load and 35 °C outdoor).)
- ε: heat exchanger effectiveness; ε_A refers to the reference point used to fix UA, in that case point A (100% load and 35 °C outdoor temperature); this is a constant for all 4 points simulated.

UA is variable and varies proportionally to the airflow rate, while NUT is supposed constant whatever the testing point[1]. And with:

$$T_{a,o} = T_{a,i} - \frac{Q_{evap}}{mCp}$$

In both cases superheat is not considered in the heat exchanger calculation, evaporating side is considered isothermal. Refrigerant fluid pressure losses are not considered either.

Condensation temperature (T_{cond}) in cooling mode:

The same iterative method on DTLM is applied at condenser as in CASE 2 for the evaporator. Condenser heat capacity for the specific point is the sum of the cooling capacity and of the compressor electricity consumption computed below so that there is an iteration on the condensing temperature value T_{cond}.
Sub-cooling and superheat horn[2] are not considered in $DTLM_2$ calculation (formula above in section CASE 2) at the condenser; thus, condenser refrigerant temperature is supposed to be constant and equals T_{cond} value.

[1] This might lead to slightly underestimate the UA value at lower air flow as U is in first order proportional to the air speed v in power of 0.75 to 0.8 and so NUT (UA/mCp) should be proportional to $v^{-0.2}$ and so slightly increases with decreased air flow. However, the refrigerant side conduction coefficient also decreases with more complex effect to model. So this simplification is considered an acceptable first order estimate and allows the model to correctly fit part load performances.
[2] Superheat horn means the transformation occurring in the condenser during which the refrigerant fluid at high temperature and high pressure flowing out of the compressor is cooled down to high pressure saturation temperature.

Evaporating temperature (T_{Evap}) in heating mode:

Evaporator capacity is the difference between the heating capacity and the compressor power. The DTLM iteration is used to compute the evaporating temperature.

Condensing temperature (T_{cond}) in heating mode:

The calculation is the same as for condensing temperature in cooling mode except the power consumption of the fan is added to the heating capacity and that the capacity of the heat exchanger is defined by the heating part load ratio of the test point simulated.

2.2.2 Evaporator superheat and condenser Subcooling estimates

Evaporator superheat (SH):

It is fixed constant to 2 K (electronic expansion valve) for all simulations. As it only intervenes in the model in modifying the compressor work, its variation has very limited impact on the global efficiency (less than 0.5 % when changing from 2 to 6 K).

Condenser Subcooling (SC):

A reference value is defined in standard rating conditions in cooling mode and at declared unit capacity at -10 °C in heating mode. In part load, Subcooling value is supposed equal to the product of the reference Subcooling value multiplied by the ratio of the specific test point temperature difference between the condensing temperature and the inlet air temperature to the same ratio for the reference test conditions (Approximation suggested by European manufacturers to model air cooled chiller SEPR performance point in the frame of Lot 1 commercial refrigeration impact assessment study [3]). For example, in cooling mode:

$$SC_{B,C,D} = SC_A \frac{T^c_{B,C,D} - T^{a,i}_{B,C,D}}{T^c_A - T^{a,i}_A}$$

where:

- $SC_{A,B,C,D}$: Subcooling in test conditions A or B or C or D in K
- $T^c_{A,B,C,D}$: condensing temperature in test conditions A or B or C or D in K
- $T^{a,i}_{A,B,C,D}$: condenser inlet air temperature in test conditions A or B or C or D in K

2.2.3 Air flow reduction at low loads for split units (with inverter compressor and fans)

At low loads in cooling mode, condenser fan power is reduced to maintain performance. This is also the case at low loads in heating mode at the evaporator. In these conditions, compressor power is low and fan power is no longer small in comparison to compressor electricity consumption; so, it is more efficient to decrease fan power, even if compressor power increases.
Refrigerant temperature is found with the same iteration of CASE 2 for evaporation temperature in cooling mode above, with changed flow rate. UA is assumed proportional to flow and NUT is constant as discussed above.
Fan power is assumed to vary with flow rate as follows for these test conditions with reduced air flow rates:

$$\mathrm{P}_{fan} = \mathrm{P}_{fan,N} \times \left(0.1 + 0.9 \times AFR^3\right)$$

where:

- AFR : ratio of the reduced to the nominal air flow
- $P_{fan,\,N}$: nominal electric power of the fan motor

2.2.4 Compressor efficiency estimate

EER (respectively COP) of the compressor is calculated from T_{Evap} and T_{Cond}. Superheat and Subcooling are also considered. A correlation between the global efficiency of the compressor and the compression ratio P_{cond}/P_{evap} is used:

$$\eta_g = f\left(P_{cond} / P_{evap} \right)$$

The compressor efficiency is then computed as the ratio of the W_{is} (isentropic compression work of the isentropic cycle) obtained for the isentropic cycle defined by T_{Evap}, SH, T_{Cond} and SC, an isentropic compression and an isenthalpic expansion. The properties of the fluid at the different state points can be computed using Refprop 9.1 [4].

$$\eta_g = w_{is} / w$$

Performance curves of a 2.9 EER (ASHRAE standard conditions, SI units) AC rotary compressor were published recently in the frame of the AHRI Low-GWP Alternative Refrigerants Evaluation Program [5]. This curve is used to model the AC rotary compressor. Note that these measurements were performed at neutral ambient around the compressor while in real life temperature surrounding the compressor may be closer to outdoor temperature. Compressor shell heat losses have been neglected in this simplified model however, and are probably not negligible at low ambient temperature.

For DC inverter rotary compressor, this curve has been corrected by the AC motor efficiency following information published by [6]. DC motor losses are supposed constant so that this performance curve is simply adjusted using a constant correction coefficient required to reach the different EER levels:

- EER of 3.15 which is the reference for average split product.
- EER of 3.4 to reach best DC inverter rotary compressor.
- For some of the units investigated in the frame of the review study [7], the compression ratio seems can be lowered down to 1.1, while it is limited to 1.2 for most DC inverter compressors.

The curves of the different rotary compressor efficiency curves (η_g) are given in Figure 1. η_g values close to nominal ASHRAE condition values can be read at compression ratio of 3.4 on the different curves, with η_g values ranging from 0.61 (EER 2.7) to 0.77 (EER 3.4).
The impact of frequency variation on compressor efficiency is not included, by lack of data.

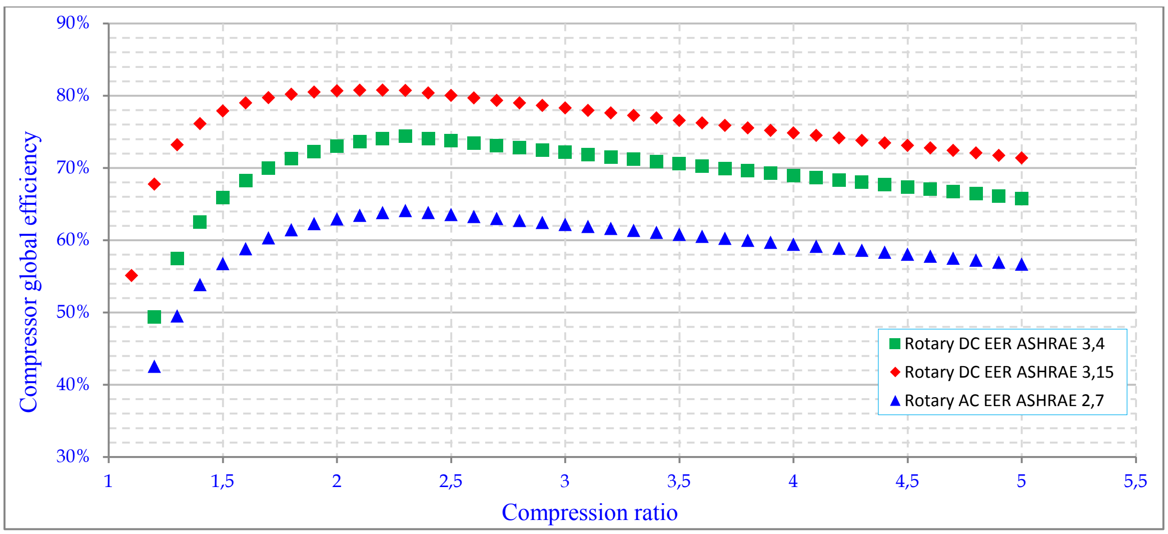

Figure 1: Compressor efficiency curve as a function of the compression ratio for the different compressor options

2.2.5 EER, SEER, COP and SCOP calculation

EER and COP are then corrected for:

- Thermostat-off, crankcase, standby power and power required for electronics (controls when unit is on)
- Fan power; Frost/defrost cycles: 5 % decrease in COP at 2 °C

3. MODEL FITTING AND SIMULATIONS

3.1 Model fitting with the base case and best available product in the European market

It is necessary to fit the model with the available known manufacturer's data. To compute the EER values, a number of parameter is already known, as Cooling capacities (load and declared values), indoor/outdoor airflow rate, power indoor/outdoor fan, air inlet temperature (entering evaporator), ambient temperature, subcooling temperature, NUT (for evaporator and condenser), and of course EER value for all working points. Once these parameters are fixed, some other parameters remain to be adjusted to get the right EER value for the full load point (A). These parameters are: the evaporator pinch value (air leaving temperature minus evaporation temperature), the condenser pinch value (condensing temperature minus outlet air temperature) and the heat exchangers effectiveness, for the other points (B, C, D) it is assumed that the NUT value is constant. Airflow rate reduction is added to the fitted parameters for the other points (B, C, D).

The COP adjustment is similar, since for reversible unit the condenser in heating mode becomes the evaporator in cooling mode with the same design parameters that must be respected when fitting heating parameters.

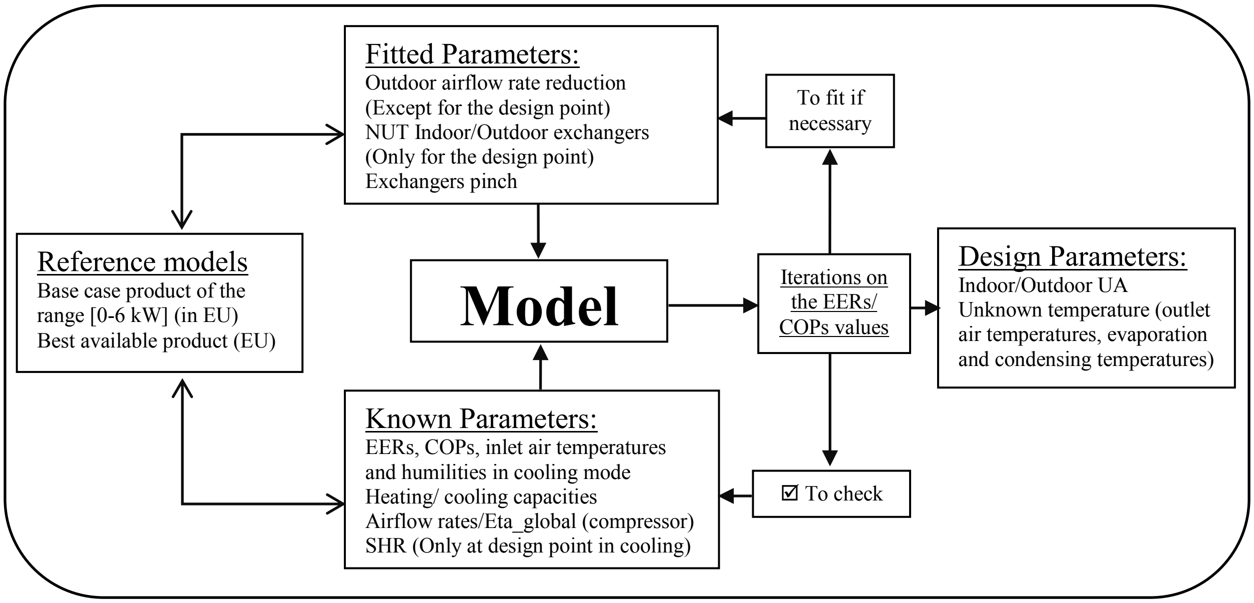

Figure 2: logigram of the fitting of the model with reference models and design parameters

To ensure that all simulations (for the average product with improvement options) done for the present study are consistent, two real products were chosen to adjust the model, a base case (average EU product, performance parameters are selected as close as possible to real products, database capacities are filtered close to the average product) and the best product available on the European market, for [0-6 kW] range of cooling capacity. For these products the cooling/heating capacities, EER/COP (at all tested points), SEER/SCOP, indoor/outdoor airflow rates are known. All simulations of products in this study (average products with improvement options) have an energy performance between the average and the best available products. If the best product efficiency level is compatible with supposed efficiency improvements, then it is likely that all simulations of products in between these efficiency levels are realistic.

The model requires knowing enough design parameters of a product; otherwise, it is difficult to predict the behavior of the model if there are more parameters to adjust than the number of equations presented above. Table below presents the main characteristics of products chosen to calibrate the model in this study:

Table 3: Air conditioning average products and best product for the range of [0-6 kW] main characteristics

	Average Product in [0-6kW]	**Best Available Product 3.5 kW**
Type / Mounting	Reversible split /wall single	Reversible split /wall single
Refrigerant Type / charge	R410A / 1.05 kg	R-32 / 1.1 kg
Cooling capacity kW (EN14511)	3.5 kW	3.5 kW
SEER[3]	6.25	10.0
EER_A/ EER_B /EER_C /EER_D[3]	3.3/5.1/8.1/11.6	4.9/7.4/12.3/20.3
PcA/ PcB/ PcC/ PcD [3] [kW]	3.5/2.6/1.7/1.2	3.5/2.6/1.7/1.2
$P_{designH}$ kW (EN14511)	3.0 kW (at -7°C)	4.1 kW (at -10°C)
SCOP[3]	4.1	5.9
COP_A/ COP_B /COP_C /COP_D	2.7/4.1/5.3/6.4	3.8/ 5.7/7.8/10.0
COP/Ph Air at 2°C and part load	2.7/1.6/1.1/1.1	2.7/1.6/1.1/1.1
T_{ol} and COP/P_H at T_{ol}	-15 °C / COP 2.3/2.5 kW	-10 °C / COP 2.7/4.2 kW
T_{biv} and COP/Ph at T_{biv}	-7 °C / COP 2.7/2.7 kW	-10 °C / COP 2.7/4.2 kW
Crankcase / Thermostat-off / Standby	3.3 W / 18 W / 3 W	0 W / 23 W / 1 W

[3] climate and load curve Regulation 206/2012

Four variables were used to quantify the overall performance of the unit: cooling/heating capacity, total electrical power, SEER and SCOP. Table 4 shows most parameters used to design the unit (results from the model). Table below summarizes all performance and design parameters of the average product (wall split 3.5 kW) fitted with the best available product. Parameters are separated by unit type, outdoor and indoor units, and a section to present the necessary parameters to calculate the SEER/SCOP values.

Table 4: Results of simulations of the average product for split 3.5kW in cooling mode

Base case of split Wall (SEER)			**Cooling capacity : 3.5 kW**			
Load	Building load	kW	3.50	2.58	1.66	0.74
	Building load	%	100%	74%	47%	21%
	Refrigerating capacity min	kW	-	-	-	1.20
	Refrigerating capacity	kW	3.50	2.58	1.66	1.20
	Load corrected	%	100%	74%	47%	34%
Evaporator	Outlet air temperature	°C	14.6	17.9	21.1	22.8
	Air inlet temperature entering evaporator	°C	27	27	27	27
	Evaporation temperature	°C	9.5	12.2	17.6	20.3
	UA Evaporator	W/K	0.28	0.28	0.28	0.28
	mCp (varies with air flow and air conditions)	kW/K	0.19	0.19	0.19	0.20
	NUT (Epsilon 0.76) = Cte	-	1.43	1.43	1.43	1.43
	Indoor air flow	m^3/h	600	600	600	600
	Power indoor fan	kW	0.03	0.03	0.03	0.03
	SHR (Sensible Heat Ratio)	%	70%	79%	100%	100%
Condenser	Ambient temperature (inlet condenser)	°C	35	30	25	20
	Condensing temperature	°C	51.7	41.1	32.3	27.0
	Liquid Subcooling	K	4.0	2.7	1.7	1.7
	UA condenser	W/K	0.40	0.40	0.37	0.27
	mCp (varies with air flow and air conditions)	kW/K	0.47	0.48	0.44	0.32
	NUT (Epsilon 0.57) = Cte		0.84	0.84	0.84	0.84
	Air flow outdoor	m^3/h	1500	1500	1350	975
	Power outdoor fan	kW	0.035	0.035	0.026	0.012
SEER	Compression ratio	-	2.98	2.15	1.48	1.20
	EER Compressor	-	3.53	5.89	11.33	19.66
	Compressor. power input	kW	0.99	0.44	0.15	0.06
	Electronics	kW	0.003	0.003	0.003	0.003
	Total power demand	kW	1.05	0.50	0.20	0.10
	EERtotal	-	3.30	5.10	8.13	11.61
	SEER	-	**6.25**			

3.2 Results of simulations product with improvement options

In this section, different improvement options (and their combinations) for air conditioners are simulated for 3.5 kW reversible split product, by applying the LCC (Life Cycle Cost analysis), in order to find the LLCC (Least Life Cycle Cost). The improvement options presented and the cost model are summarized in table 5.

Cost model is based on the review study [7] with adjustments. Manufacturer overcosts (additional costs due to design options) are directly passed to the final end-user with the markup factors from manufacturer cost to manufacturer selling price. Concerning the price increase of heat exchanger coils, the reference is the price in the review study [7]. Having a unit with doubled capacity increases the manufacturing cost of heat exchangers by 100 %; and the price increases with power of 0.8 of the heat exchanger area increase. See the following equation:

$$Cost\_Heat\_exchanger = A + B \times (UA^{0.8})$$

A and B are constants to be determined with the initial cost and its double at 100% increase of UA. The coefficient 0.8 gives higher price for large increase than for smaller ones, which is coherent with the larger adaptation requirements (for instance casing size change, fan size change).

In addition, the cost of the fan (larger fan), the cost of the refrigerant fluid mass used and the cost of the casing (bigger size) also vary. For these components, the same method is applied as for heat exchangers. These costs are shared between indoor and outdoor units with a respective prorate of 45% and 55%. The cost of microchannel heat exchanger is 1.3 times the cost of Cu-Al composed tube and fin (of the outdoor unit).

Table 5: cost model for a 3.5 kW unit (6.25 SEER) and improvement options tested

Cost model for 3.5 kW of average product	
Compressor	18%
Condenser	18%
Evaporator	12%
Outdoor fan	9%
Indoor fan	6%
Working fluid	4%
Refrigerant line	6%
Controller + Electronics	6%

Improvement options	
Option CP1	Rotary compressor 3.4 EER
Option CP2	Rotary compressor 3.4 EER w improved oil management
Option HE1	UA value of indoor heat exchanger increased by 40 %
Option HE2	UA value of indoor heat exchanger increased by 80 %
Option HE3	UA value of outdoor heat exchanger increased by 40 %
Option HE4	UA value of outdoor heat exchanger increased by 80 %
Option LPM	Lowest values achievable for SB and TO
Option MHE	Microchannel heat exchangers for the outdoor unit

Hypothesis for LCC calculation are:

- Life time: 12 years; Installation cost: 800 Euros; Maintenance: 4% of the initial investment
  PWF = life time, as discount rate of 4 % equals to escalation rate of energy prices
- Electricity price: 0.195 €/kWh for 0-6 kW units and 0.187 €/kWh for 6-12 kW units.
- Heating hours : 1400 hours / Cooling hours : 350 hours

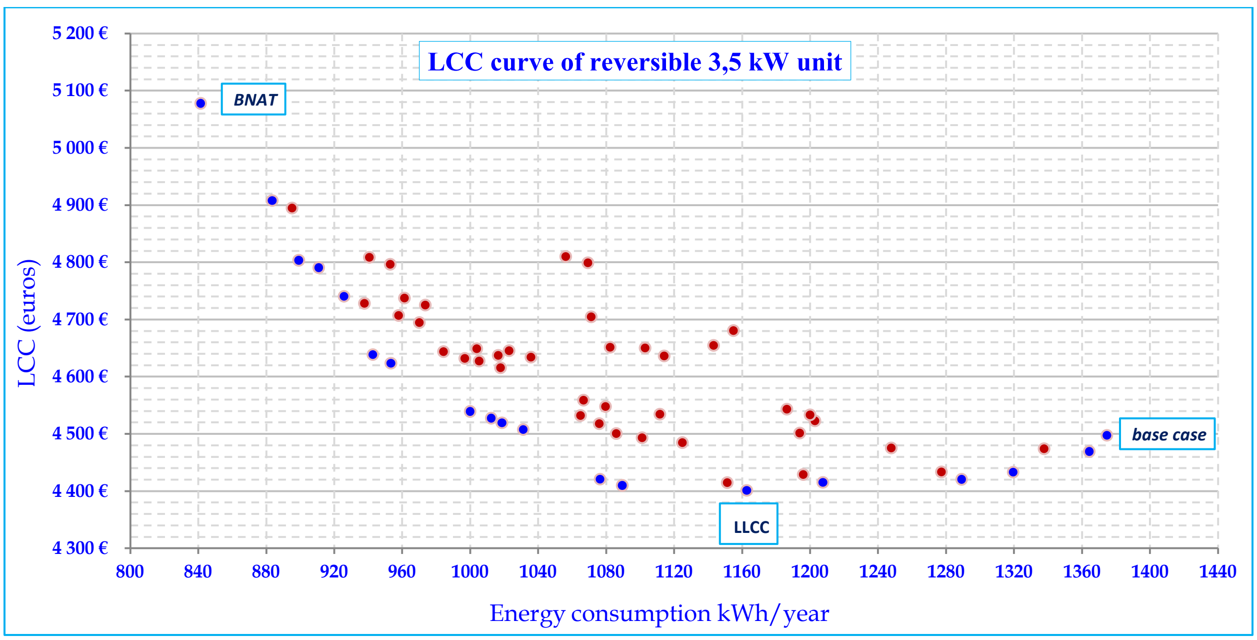

Figure 3: LCC curve of 3.5 kW unit

Improvement impact has been computed for a large number of possible combinations. The lowest LCC value at a given energy consumption level (bottom line of all LCC points) are presented in blue in figure 2, BNAT (best not available technology) option is the combination of all improvement options together. This analysis allowed having an idea about the most profitable improvement combined options.

The LLCC for this product is then the option HE1 with:

- SEER 6.7 / SCOP 4.4 with an Energy consumption decrease is about 10 % over the average product.
- LCC = 4400 €, 7 years of payback time

BNAT (with all options): SEER= 11.4, SCOP = 5.9, LCC= 5080 €, 15 years of payback time

Table 6 shows the relative impact of the improvement options on the SEER and SCOP of the average product the European market:

Table 6: SEER and SCOP increase with the evaporator/ condenser UA increase

Improvement option	% SEER	% SCOP
40% increase of evaporator UA	14%	10%
80% increase of evaporator UA	20%	17%
40% increase of condenser UA	12%	6%
80% increase of condenser UA	20%	10%

Note this is not the final LLCC value proposed in the review study [7], as electricity prices have been changed to establish final figures.

To conclude, the methodology is proposed to use the minimum model parameters and variables to compute the performance indicators and energy consumption in as simple form as is possible. In the model, only influencer terms are included in expressions.

4. CONCLUSIONS

The objective of the work reported here was to present a simplified model able to quantify energy gains with improvement options for air conditioners and air-to-air heat pumps. Real products were used to adjust the model; one of limitations of the model is the compressor curve, which remains inaccessible for most DC inverter rotary compressors in the market.

The simplicity of the model is based on the use of simple software like MS excel with thermophysical properties database to compute a complicated thermodynamic system (the use of compressor curve with temperatures instead of pressures would allow to automate the model with only MS excel).

The simulations seem to give results not far from what announces product designers ; though, two major uncontrolled sources of error need to be further investigated, the first one regards the uncertain compressor performance (performance curve itself including impact of frequency and the impact of heat losses through compressor envelope), the second one regards the heat exchangers (very simplified model - semi-isothermal for both evaporator and condenser, and pressure losses on the refrigerant).. Somehow, these errors are probably partly compensating in the model.

For the LCC analysis in this European study, modeling several points and products with complex models was not possible in the limited timeframe, so the present model was considered as an acceptable compromise to evaluate the impact of the improvement options on the seasonal indicators of energy performance. However, it is planned to further compare this simplified evaluation tool to a more detailed model to ensure the consistency of the behavior of certain parameters under all conditions

5. REFERENCES

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